NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE

No. 1537

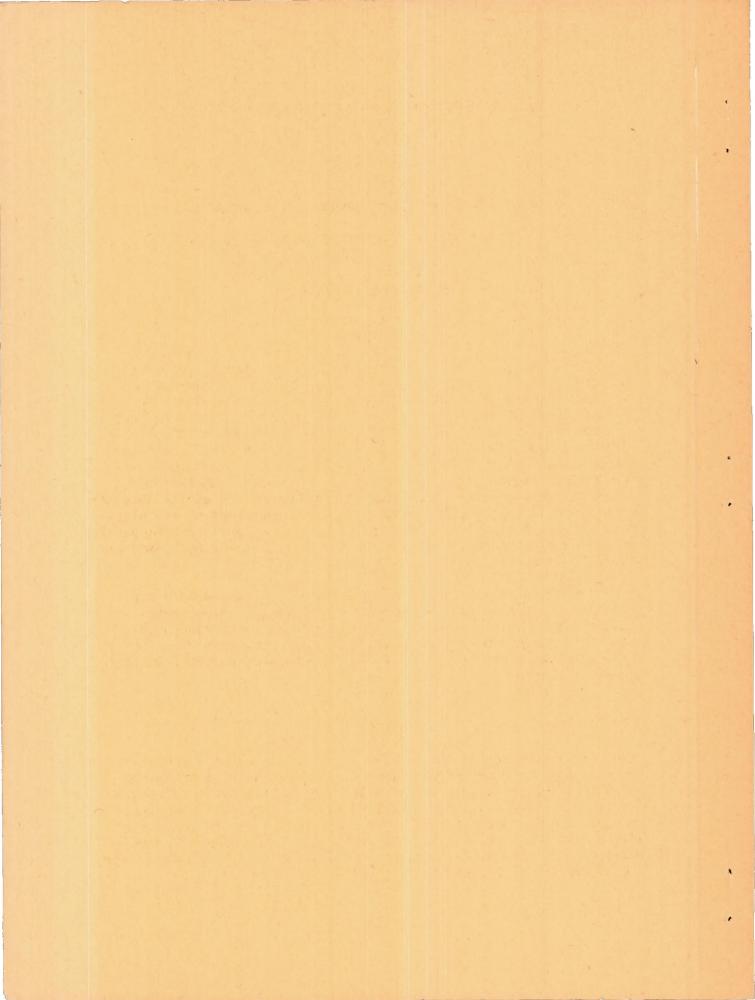APPARENT EFFECT OF INLET TEMPERATURE ON ADIABATIC

EFFICIENCY OF CENTRIFUGAL COMPRESSORS

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SUMMARY

An investigation was made to determine whether the apparent
variation of adiabatic efficiency with inlet temperature for cen-
trifugal compressors could be reduced or eliminated. The effects
of heat transfer and Reynolds number on adiabatic efficiency were
considered. Three centrifugal compressors were tested in an
insulated variable-component compressor at several inlet tempera-
tures with constant discharge pressure and at two discharge pres-
sures with constant inlet temperature. Discharge-temperature
measurements were made at the diffuser discharge and at the entrance
to the discharge pipe, in addition to the measurement at the stan-
dard station in the discharge pipe, to determine whether a temperature-
measuring station could be found where the effects of heat transfer
would be less than those at the standard station.

The standard methods of determining centrifugal-compressor
efficiency by temperature-rise measurement tend to give erron-
eously high values when ambient room-air inlet temperatures are
used, and the apparent variation of adiabatic efficiency can be
eliminated by basing efficiency on temperature measured at the dif-
fuser discharge.

INTRODUCTION

Investigations of the performance of centrifugal compressors
at the NACA and other laboratories have shown that the adiabatic
efficiency apparently increased with inlet temperature at constant
equivalent tip speeds and equivalent volume flows, but that the
pressure ratio is unaffected. This apparent change in efficiency
has been shown to be related to heat transfer (reference 1) but no
adequate nor complete explanation has been available to account for
the variation of efficiency with inlet temperature.

The lack of agreement of efficiencies determined for the same compressor at the same equivalent tip speeds but at different inlet temperatures renders the interpretation of compressor performance incomplete. The uncertainties that arise from the variation in temperature-rise-ratio efficiency are:

(a) Inconsistencies in efficiency in compressor investigations conducted with different inlet-air temperatures but at corresponding equivalent tip speeds and with the same procedures

(b) Inability to determine precisely the performance of a compressor from performance data of the same compressor obtained at a different inlet-air temperature

(c) Inability to determine the true efficiency of a compressor from temperature-rise measurements without a determination of the effect of inlet-air temperature and the probability that efficiencies obtained by standard methods at sea-level atmospheric temperature have been higher than the true value

In an investigation of the effect of blade curvature on centrifugal-impeller performance (reference 2), efficiencies based on measurements made according to standard procedures were found to be inconsistent when different inlet temperatures were used. The pressure ratio was practically unaffected by variations in inlet temperature, whereas the temperature-rise ratio was affected. The basic parameters of equivalent tip speed and equivalent volume flow were the same at all points of comparison, but the Reynolds number varied because the compressor-discharge pressure was maintained constant regardless of inlet temperature. Heat-transfer effects might have been responsible for the lack of agreement in temperature-rise ratios at various inlet temperatures, and the variation of Reynolds number with inlet temperature might account for a disagreement of pressure ratio, temperature-rise ratio, or both.

An investigation was made at the NACA Cleveland laboratory to determine whether the apparent variation of adiabatic efficiency with inlet temperature could be reduced or eliminated. In order to reduce the effect of heat transfer on the discharge-air temperature, temperature measurements were made at the diffuser discharge and at the entrance to the discharge pipe in addition to the standard measurement in the discharge pipe. Adiabatic efficiencies based on discharge temperatures taken at the diffuser discharge and at the standard station in the discharge pipe 12 diameters downstream of the collector are presented for three impellers at several equivalent tip speeds and inlet temperatures. The results of varying only Reynolds number for several equivalent tip speeds are presented and probable reasons for the apparent variation of adiabatic efficiency with inlet temperature are discussed.

APPARATUS AND PROCEDURE

Setup

Three centrifugal impellers, designated A, B, and C, and a variable-component compressor set up in accordance with the recommendation of reference 3 were used for this investigation. A schematic cross section of the variable-component compressor is shown in figure 1. A typical installation of one of the impellers, the rear half of the vaneless diffuser, the collector, and other component parts is shown in figure 2. The air discharged from the vaneless diffuser flows into the collector chamber before entering two tangential discharge pipes located 180° apart. The variable-component compressor was mounted separate from the driving gearbox to eliminate direct heat conduction between the compressor and the gearbox. The compressor was completely enclosed in an insulation box. The walls of the insulation box were composed of 1/4 inch of hard asbestos, 1 inch of insulating board, and 1/2 inch of plywood. The inlet and discharge pipes were insulated with 3 inches of mineral wool.

Instrumentation

The compressor was instrumented in accordance with the recommendations of reference 3 with static-pressure taps, total-pressure tubes, and exposed-bulb thermocouples. This instrumentation provides an inlet measuring station 2 pipe diameters upstream of the impeller and a discharge measuring station in each discharge pipe 12 pipe diameters downstream of the collector. Pressure measurements were also made at the diffuser discharge to determine the pressure ratio across the compressor up to this point.

In addition to the standard thermocouples, the following thermocouples were installed on the compressor:

(1) Two high-recovery-factor thermocouples were placed diametrically opposite each other and midway between the front and rear diffuser walls at the diffuser discharge for air-temperature measurements. These thermocouples were shielded and were similar to the Pratt & Whitney probe described in reference 4. The diffuser thermocouple locations are shown in figure 2.

(2) One exposed-bulb thermocouple, the same as those used at the standard measuring station, was placed in the entrance to one of the discharge pipes.

The impeller speed was measured with an automatically timed electric revolution counter. The precision of measurements is estimated to be within the following limits:

Temperature, $^{\circ}$R . . . . ±0.5
Pressure, in. Hg . . . . ± 0.02
Impeller tip speed, percent . . . . ±0.5
Volume flow, percent . . . . ±2.0

Procedure

Impellers A, B, and C were run at a number of inlet temperatures and equivalent (corrected) tip speeds (reference 5) with the discharge temperature measured at the standard discharge station and at the diffuser discharge for all three impellers. Air temperature was also measured at the entrance to the discharge pipe for impellers B and C. The following range of equivalent tip speeds and inlet temperatures was investigated:

Equivalent tip speed (ft/sec)	Inlet temperature, $^{\circ}$R		
	Impeller A	Impeller B	Impeller C
900	400, 535	463, 530	452, 534
1100	403, 545	399, 452, 526	431, 531
1300	400, 548	402, 529	457, 531

Impeller A was also run at equivalent tip speeds of 879, 1178, and 1350 feet per second at discharge pressures of 40 and 20 inches of mercury absolute to obtain an estimate of the possible effect of Reynolds-number variation on adiabatic efficiency. The inlet temperature was nearly the same for both discharge pressures, and the discharge temperatures were taken at only the standard station in the discharge pipe.

Calculations

The adiabatic efficiency, based on the pressure ratio across the compressor up to the diffuser discharge and the temperatures at the standard discharge-pipe measuring station, was calculated for all runs. Adiabatic efficiency was also calculated from the same pressure ratio, but for the temperatures at the diffuser discharge and at the discharge-pipe entrance. The difference between the adiabatic efficiencies for any run is therefore due to the difference in temperature-rise ratios.

The Reynolds number R is defined as

$$R = \frac{D\rho_1 U}{\mu_1}$$

where

D   impeller diameter, feet

ρ_1 inlet stagnation density, slugs per cubic foot

U   impeller tip speed, feet per second

μ_1 absolute viscosity at inlet stagnation conditions, slugs per
      foot-second

This Reynolds-number determination has value only in comparing points at the same equivalent volume flow and equivalent tip speed for different inlet or discharge conditions of temperature or pressure.

RESULTS AND DISCUSSION

Apparent Effect of Inlet Temperature on Adiabatic Efficiency

The apparent variation of adiabatic efficiency with inlet temperature was examined by comparing efficiencies based on temperature measurements at the standard discharge-pipe station, at the entrance to the discharge pipe, and at the diffuser discharge.

Adiabatic efficiency based on discharge-pipe temperatures. - The adiabatic efficiency η_{ad} is presented for three equivalent tip speeds $U/\sqrt{\theta}$ (θ, ratio of inlet temperature to NACA standard sea-level temperature, 518.4° R) of impellers A, B, and C in figures 3, 4, and 5, respectively. The efficiencies based on the temperatures measured at the standard measuring stations in the discharge pipes are shown for impellers A, B, and C in figures 3(a), 4(a), and 5(a), respectively. These figures show that at a given equivalent tip speed and equivalent volume flow $Q_{t,1}/\sqrt{\theta}$ (where $Q_{t,1}$ is inlet-volume flow at stagnation pressure), the adiabatic efficiency apparently increased with inlet temperature. The efficiency of impeller B (fig. 4(a)) at an equivalent tip speed of 1100 feet per second was reduced about 0.02 when the inlet temperature was reduced from 526° to 452° R, and about 0.04 for an inlet-temperature reduction from 526° to 399° R. The same order of magnitude of efficiency variation for similar changes in inlet temperature are shown for impeller A (fig. 3(a)) and impeller C (fig. 5(a)). The temperatures measured at the

entrance of the discharge pipe were the same as those at the standard
station; any heat loss from the discharge pipes was therefore too
small to make a measurable temperature drop between these thermocouple
locations, and the adiabatic efficiency was the same as at the stan-
dard station in the discharge pipes.

Adiabatic efficiency based on diffuser-discharge temperature. -
The adiabatic efficiency based on the temperature measured at the
diffuser discharge is presented for impeller A, B, and C in figures 3(b),
4(b), and 5(b), respectively. For each of the three impellers, the
efficiency, based on diffuser-discharge temperature, was practically
unaffected by variation of the inlet-air temperature. Impeller B
(fig. 4(b)) shows complete agreement of efficiency and little variation
in volume flow. For impellers A and C, there are differences in
efficiency and volume flow that are within the range of precision
of the measurements; therefore, it is indefinite whether or not small
actual differences do exist. The adiabatic efficiency based on
diffuser-discharge temperature remains substantially constant with
change in inlet-air temperature for the three impellers used in the
variable-component compressor.

Effect of Reynolds Number on Adiabatic Efficiency

The adiabatic efficiency of impeller A is shown in figure 6 for
discharge pressures of 20 and 40 inches of mercury absolute at three
equivalent tip speeds and nearly constant inlet temperature. At a
given equivalent volume flow, the Reynolds number for the 20-inch
discharge pressure is approximately one-half the value for the 40-inch
discharge pressure.

At equivalent tip speeds of 879 and 1178 feet per second, some
small disagreements existed between the adiabatic-efficiency curves at
the two discharge pressures (figs. 6(a) and 6(b), respectively) but
almost perfect agreement was obtained at 1350 feet per second (fig. 6(c)).
The variation of the discharge pressure from 20 to 40 inches of mercury
absolute at constant inlet temperature caused a change in Reynolds num-
ber one and one-half times that obtained by the greatest variation of
inlet temperature at constant discharge pressure. Although changes in
the shape of the adiabatic-efficiency curves and in maximum equivalent
volume flow resulted from a change in Reynolds number, these changes were
too small and too inconsistent to account for the variation of adiabatic
efficiency with inlet temperature at constant discharge pressure.

Discussion of Inlet-Temperature Effect

The elimination of the apparent effect of inlet-air temperature on
adiabatic efficiency when discharge temperature is measured at the diffuser
discharge and the lack of effect of inlet-temperature variation on the

pressure relation suggest that the effect is due to heat transfer from the compressor to the atmosphere between the diffuser discharge and the discharge pipes. The difference between the diffuser-discharge temperature and the temperature in the discharge pipe bore a substantially linear relation to the temperature difference between the diffuser discharge and ambient room air. The heat losses from the variable-component compressor that would be required to account for the differences in efficiencies based on the temperatures at the diffuser discharge and in the discharge pipe are shown in the following table. The values shown apply to data points from figure 3 at the tip speed of 1100 feet per second; the heat loss would occur through the collector (figs. 1 and 2), which is approximately 41 inches in diameter by 14 inches maximum depth.

Equivalent volume flow $Q_{t,1}/\sqrt{\theta}$	Efficiency at diffuser discharge η_{ad}	Inlet temperature ($^{\circ}$R)	Room temperature ($^{\circ}$R)	Diffuser-discharge temperature minus room temperature ($^{\circ}$F)	Temperature drop, diffuser to discharge pipe ($^{\circ}$F)	Required heat loss (Btu/sec)
6690	0.648	545	545	175.8	6.6	9.8
5817	.780			178.7	6.3	7.0
4400	.732			188.3	7.7	6.5
6583	0.641	403	545	-6.0	0.4	0.7
5563	.772			-11.5	-.8	-1.0
4672	.753			-6.9	-.3	-.3

These indicated heat-transfer quantities have not been satisfactorily accounted for by computations using coefficients for uniform flows.

Although the explanation of the nature of the effect of inlet temperature on adiabatic efficiency is incomplete, it was found that if the efficiency is based on a temperature measured where external heat transfer has had negligible opportunity to occur, the apparent variation in adiabatic efficiency is eliminated. It is also apparent that the efficiencies which vary most from the correlation values are for the highest inlet temperature and that these efficiencies are higher than the correlation values.

Centrifugal-compressor efficiency ratings obtained by standard pro-
cedures with ambient room air inlet temperature thus tend to be
higher than the true efficiency.

A method of obtaining consistent efficiencies for comparative
performance in a variable-component centrifugal compressor has
been obtained; however, these efficiencies are not necessarily
true efficiencies free from external effects. Obviously, the true
values of adiabatic efficiency can be found only by obtaining a
complete heat balance for each compressor or by use of an accurate
dynamometer.

CONCLUSIONS

From an investigation of the apparent effect of inlet-air tem-
perature on adiabatic efficiency conducted in a variable-component
compressor, the following conclusions were drawn:

1. The apparent variation of adiabatic efficiency with inlet
temperature of centrifugal compressors in a variable-component com-
pressor of the type used in this investigation can be eliminated by
basing the efficiency on discharge temperature measured at the
diffuser discharge.

2. The standard methods of determining centrifugal-compressor
efficiency by temperature-rise measurement tend to give erroneously
high values when ambient room-air inlet temperatures are used.

3. A reliably accurate determination of the efficiency of a centri-
fugal compressor can be made by temperature measurements only if a com-
plete heat balance of the particular compressor is obtained.

Flight Propulsion Research Laboratory,
    National Advisory Committee for Aeronautics,
        Cleveland, Ohio, June 27, 1947.

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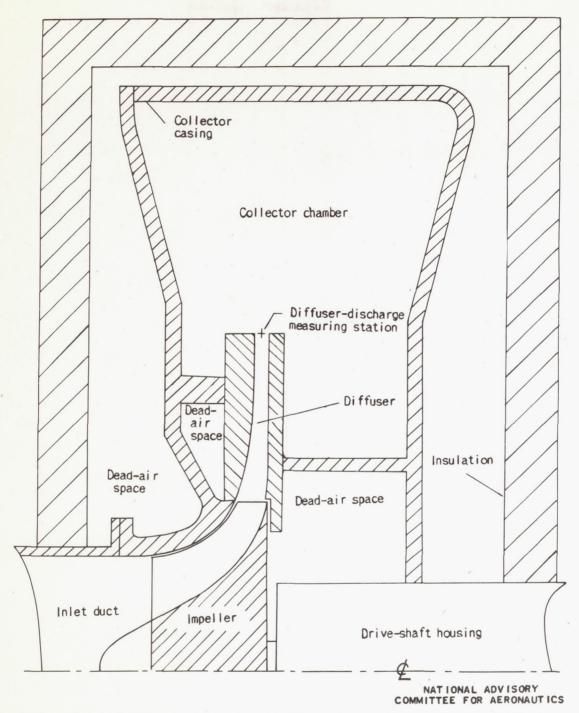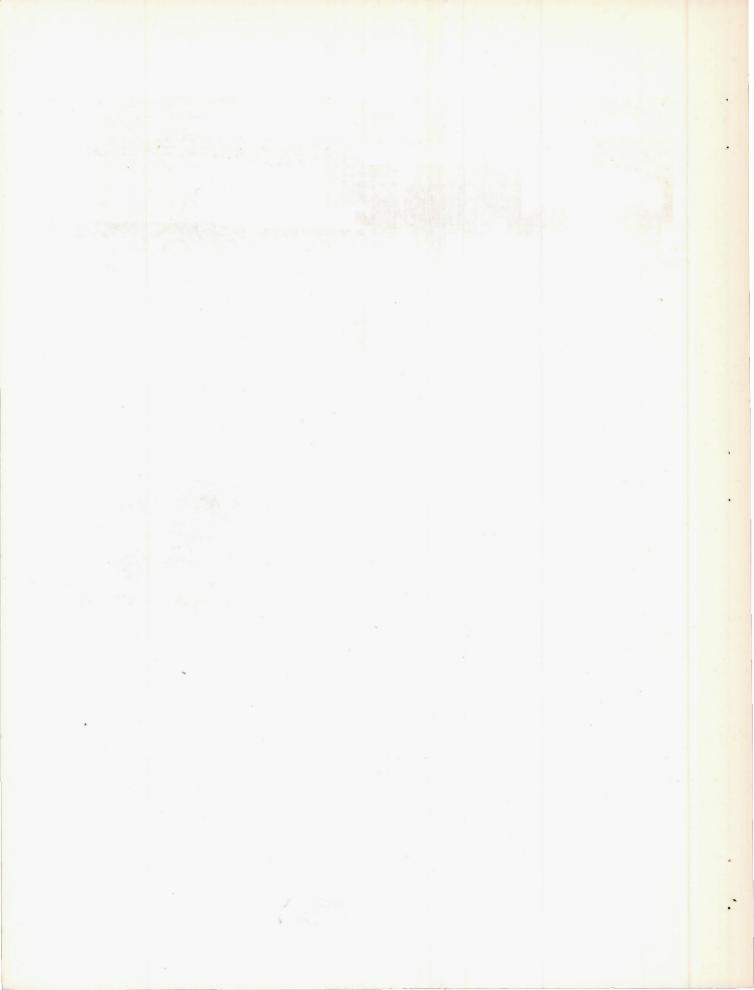   NACA ARR No. E5F13a, 1945.

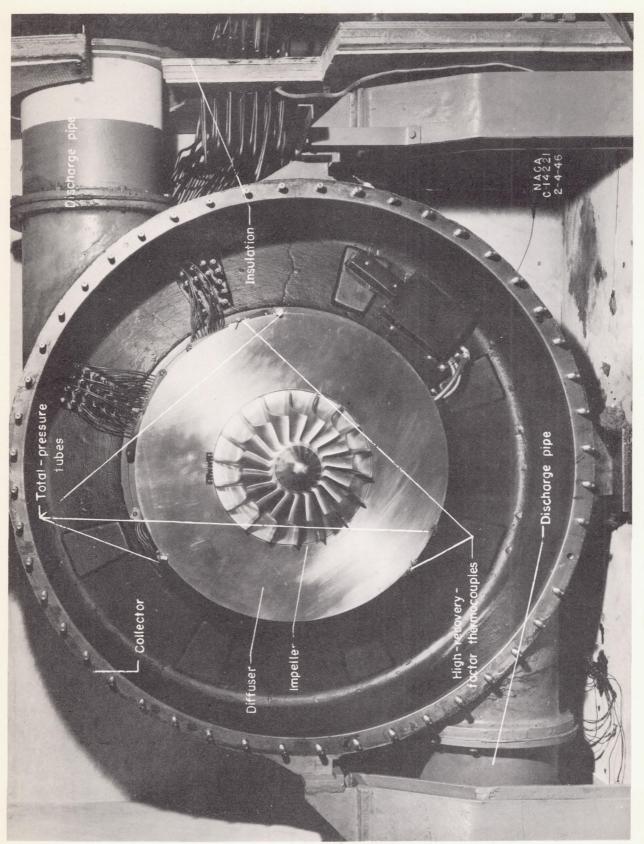Figure 1. - Cross section of variable-component compressor. (Scale 3/8 in. = 1.0 in.)
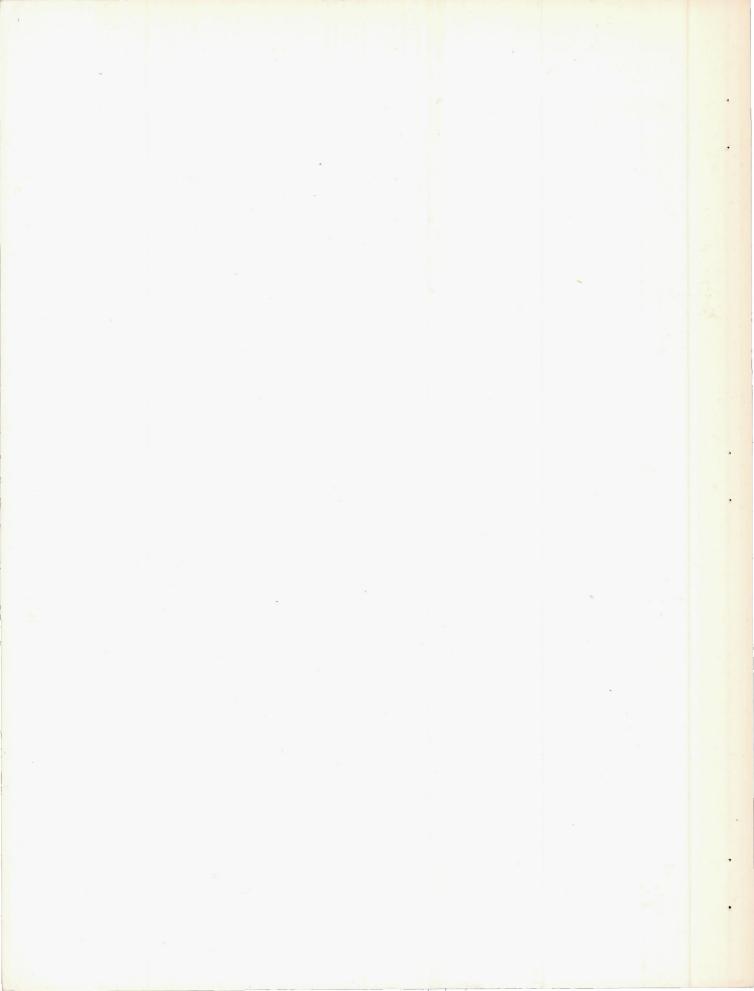
Figure 2. – Front view of variable–component compressor.

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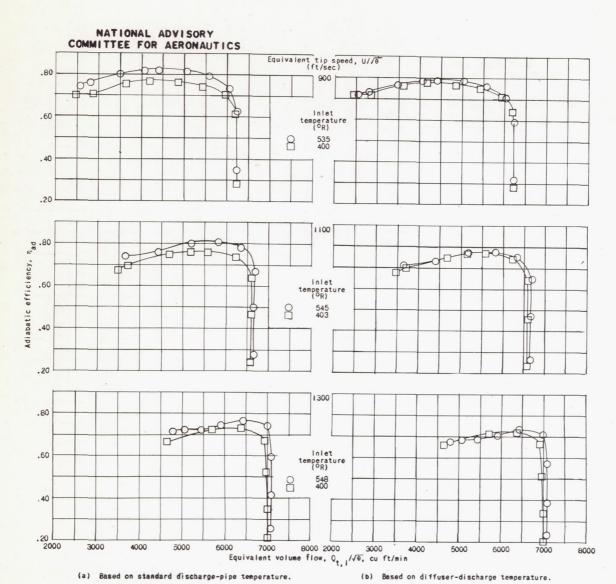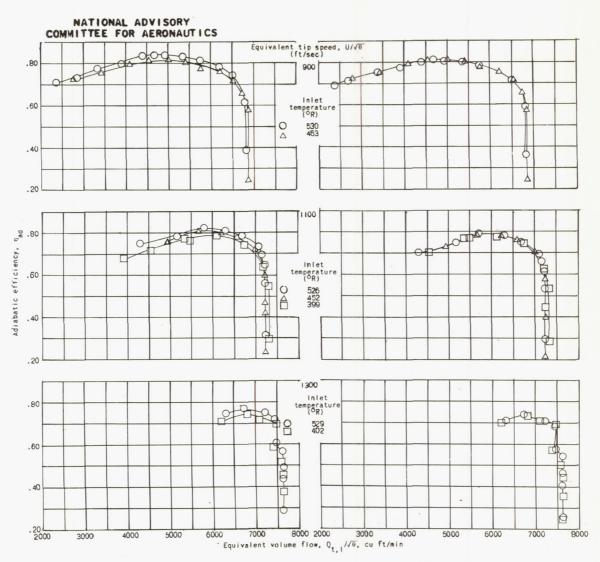COMMITTEE FOR AERONAUTICS

Figure 3. – Adiabatic efficiency of impeller A.

(a) Based on standard discharge-pipe temperature.

(b) Based on diffuser-discharge temperature.

(a) Based on standard discharge-pipe temperature.   (b) Based on diffuser-discharge temperature.

Figure 4. - Adiabatic efficiency of impeller B.

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Equivalent tip speed, U/√θ
(ft/sec)

900

Inlet
temperature
(°R)
○   534
△   452

1100

Inlet
temperature
(°R)
○   531
△   431

1300

Inlet
temperature
(°R)
○   531
△   457

Adiabatic efficiency, η_ad

Equivalent volume flow, $Q_{t,1}/\sqrt{\theta}$, cu ft/min

(a) Based on standard discharge-pipe temperature.   (b) Based on diffuser-discharge temperature.
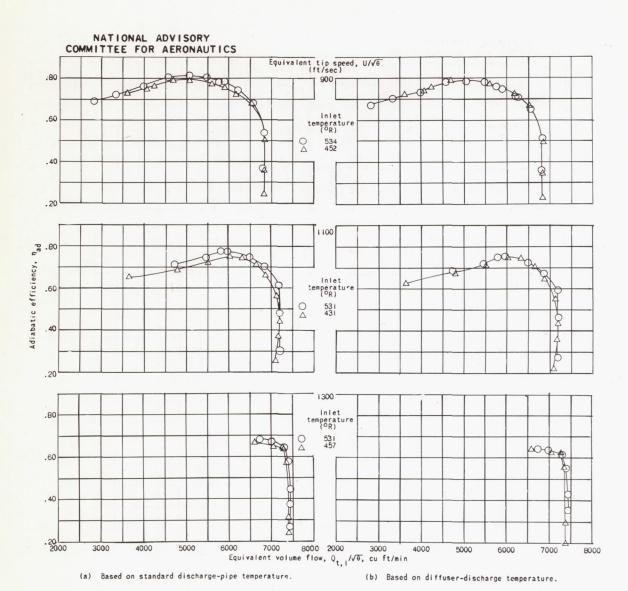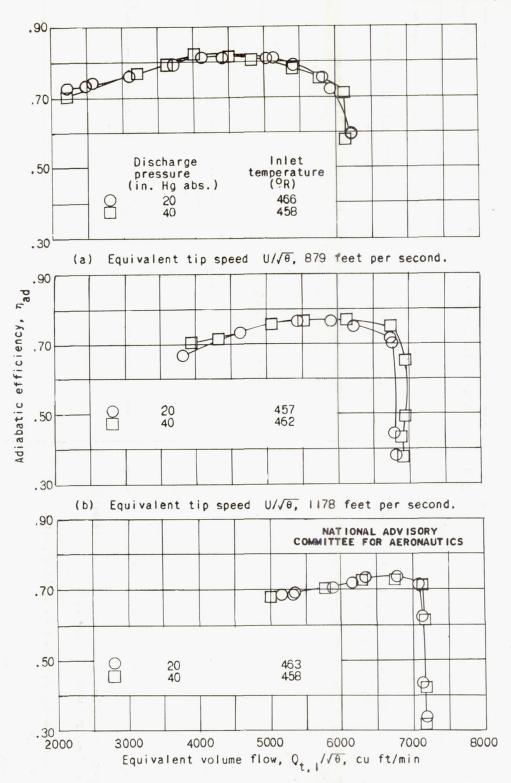
Figure 5. — Adiabatic efficiency of impeller C.

(a) Equivalent tip speed $U/\sqrt{\theta}$, 879 feet per second.

(b) Equivalent tip speed $U/\sqrt{\theta}$, 1178 feet per second.

(c) Equivalent tip speed $U/\sqrt{\theta}$, 1350 feet per second.

Figure 6. - Effect of Reynolds number on adiabatic efficiency of impel-
ler A.